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EXPERIMENTAL STUDY OF A GAS JET PENETRATING A LIQUID COOLANT AND IMPINGING ON A HEATED SURFACE

by

B. M. Hoglund, R. P. Anderson,
and L. Bova

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B. M. Hoglund,* R. P. Anderson,
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Reactor Analysis and Safety Division

March 1971

*Now with Center for Environmental Studies

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ABSTRACT

This document describes an experimental study of temperature transients induced in a heated wall as a result of a gas jet penetrating the coolant stream and impinging upon the wall. The purpose of the study was to develop insight into the mechanism of heat transfer associated with this phenomena.

The experiment was performed in a water-cooled, electrically heated, rectangular flow channel, 0.050 x 0.578 x 18 in. The test section was designed to allow a fixed volume of gas to be discharged through an orifice and impinge on the opposite wall. Thermocouples measured the resulting wall temperature transients. The test parameters were:

Coolant velocity -	10, 20, 30 fps
Orifice diameter -	0.016, 0.025, 0.051 in.
Plenum gas pressure -	500, 750, 1000 psia
Surface heat flux -	0.5×10^6, 1.0×10^6 Btu/hr-ft^2

Tests were run with and without bypass flow around the heated channel.

Additional data were obtained from an unheated Lucite test section. Transient pressures and flow rates were measured, and high-speed motion pictures of the resulting gas jets and flow transients were obtained.

It was concluded from this experiment that an impinging gas jet is unlikely to produce failure in an adjacent fuel pin at the point of impingement. In every case, the surface at the point of impingement was cooled to a temperature less than the steady-state temperature. Only for cases where the channel was blown completely dry did the wall temperature exceed its initial steady-state value.

I. INTRODUCTION

The experiment described herein is the first of several out-of-pile experiments directed toward the study of potential propagation of failure from one fuel pin to another in an LMFBR fuel subassembly. The Fuel Element Failure Propagation Program at Argonne National Laboratory does not address itself to why or how the initiating failure occurred; rather, it is designed to answer the question, "Given the failure of a fuel pin in an operating LMFBR, are there pheonmena that will induce failures in adjacent fuel pins?"

During reactor operation fission produces a number of fission gases in the fuel, xenon being the most abundant. A portion of this gas remains in the fuel; the remainder diffuses out of the fuel and is retained by the cladding. Fission gas plenums are designed into the fuel pins to limit the pressure buildup due to this fission gas production. Even so, the internal pressure may reach values as high as 1000 psi near the "end-of-life" of the fuel. This high-pressure gas represents a source of stored energy that could provide the driving force for propagation of failure once a break has occurred in the fuel-pin cladding. The release of stored fission gas has, in fact, been the potential mode of failure propagation receiving the most attention to date.

Data on the most probable mode and location of fuel-pin failures were not available at this time. In addition, there were very little data on many of the parameters that would influence the rate at which the gas escaped from the cladding breach, two of the more important ones being the flow path taken by the gas and the porosity or resistance of the fuel to the flow of fission gas. In view of the long times required to obtain this type of data it was decided that, to meet the objectives of the Fuel Element Failure Propagation program, the entire spectrum of gas release rates would be studied in out-of-pile tests to try to define those release rates that might lead to potential of propagation damage.

To study the phenomena of failure propagation resulting from a release of fission gas from a ruptured fuel pin, it was convenient to define three regimes of gas release rates. The first was identified with those cladding failures that result in a very rapid release of gas and may cause failure propagation by overheating of the adjacent pins as a result of hydraulic transients that temporarily void the region. The gas release rates are controlled primarily by the coolant inertia.

The second regime covered somewhat slower rates of gas release that produce a persistent gas jet. This jet may impinge on adjacent pins and produce local damaging hot spots and failures. The propagation potential of the flow transient and the pressure was considered secondary to the overheating produced by the gas blanketing effect of the jet.

The third regime covered still slower rates of gas release. In this regime, the postulated mode of propagation was overheating of the pin due to a local flow reduction produced by the two-phase-flow characteristics of the fission gas and sodium mixture. In this case, the gas release rate was not rapid enough to produce a high-velocity jet but released a quantity of gas that significantly alters the local flow rates of sodium. This release could last over a period of seconds and would be most damaging if it occurred near the core inlet.

This experiment was designed to study phenomena associated with the second regime, i.e., a persistent gas jet impinging upon a heated surface. Very little is known about what happens when a high-velocity gas jet penetrates a dense, high-velocity fluid flowing in a direction normal to the path of the jet. The gas jet has sufficient momentum to penetrate the fluid and, in the narrow confines of a fuel subassembly, impinge on an adjacent fuel pin. The unknowns manifest themselves in attempts to predict the transient heat-transfer effects on the heated surface. For instance, is the adjacent surface area completely gas blanketed or are fluid droplets entrained in the jet with the result that the surface is spray cooled? If the surface is usually gas blanketed, many aspects of the problem could be studied using water as the coolant; however, if the surface is usually spray cooled, then the large differences in the heat-transfer characteristics between water and sodium would require the problem to be studied in sodium.

II. EXPERIMENTAL OBJECTIVES

This experiment was intended to scope the problem of gas-jet impingement and provide some preliminary insight into the heat-transfer mechanisms associated with it. The results of these tests were needed to specify the coolant (water or sodium) in subsequent experiments.

The specific experimental objectives were the following:

1) Obtain data for the temperature transients induced in a wall by the impingement of a high-velocity gas jet.

2) Deduce information about the heat-transfer phenomena occurring when the jet impinges on the surface; i.e., is the surface truly gas blanketed or does the gas jet entrain sufficient liquid to spray-cool the surface?

3) Obtain transient flow, pressure, and photographic data to characterize the hydrodynamic phenomena associated with the gas release.

4) Investigate the effect of bypass flow around a channel subjected to a rapid gas release.

The original plan was to deduce the mechanism of heat transfer by comparing the measured surface-temperature transients with transients calculated for various assumed heat-transfer conditions. In order to obtain the required temperature measurements as rapidly as possible, an available, instrumented, electrically heated test section (left over from the AARR reactor test program) that was designed to fit into the selected experimental facility was used. The test section had two thermocouples welded directly to the outer surface of the copper heater, near the channel exit, to serve as burnout detectors. For this experiment, the test section was positioned so the thermocouples would be near the coolant inlet. It was not possible to obtain measurements of the wall temperature near the heater exit; measurements of general interest, but not required to meet the objectives of this study.

The test section was positioned with the thermocouples near the channel inlet so that the gas released from the orifice, located opposite the upper thermocouple, would travel through nearly the full length of the heated section. The object was to produce the most severe transient.

It was decided that high-speed motion pictures of the phenomena would be useful. While a gas-injection system was being installed on the heated section, a clear plastic test section was fabricated. Tests were performed for several combinations of gas and coolant flow rates while being photographed with a high-speed motion-picture camera.

III. DESCRIPTION OF EXPERIMENTS

The study consisted of two experiments. The first was performed with an unheated, Lucite test section with flow-channel dimensions that matched the dimensions of the heated test section used in the second experiment. The clear Lucite test section allowed high-speed motion pictures to be taken of the coolant flow transients and of the effect of the gas jet impinging upon the wall. Transient flow and pressure data were obtained for comparison with the flow and pressure transients measured during the tests with the heated test section.

The second test utilized an electrically heated test section, instrumented with thermocouples on the wall and pressure transducers at the inlet and exit, to measure the temperature and pressure transients produced by the sudden release of gas from variously sized orifices located in the wall of the test section. The experiment was performed using water for the coolant and nitrogen for the gas. The test section was mounted vertically in a high-pressure test facility capable of circulating the desired quantity of water through the test loop. A bypass-flow channel was placed in parallel with the test section to simulate the flow paths in a reactor core. The system pressure was maintained at 200 psi at the test-section inlet (this was required to prevent boiling at the exit during the low-flow test).

The gas-injection system consisted of a fixed-volume reservoir connected through solenoid valves to a gas-supply bottle on one end and on the other end to a short piece of tubing with an orifice in the end. The end of the tubing with the orifice was mounted flush with the inside surface of the channel wall. The reservoir was charged to the desired pressure through a charging solenoid valve. The gas release was initiated by opening the second solenoid valve.

The experimental procedure for the unheated tests consisted of the following steps:

1) The system flow rate was set to give the desired velocity (10, 20, or 30 fps) in the test section and to provide a bypass flow rate at least ten times larger than that in the test section. (Some tests were run with no bypass flow to observe the effect on the hydraulic transients of different boundary conditions at the channel inlet and exit.)

2) A series of test runs were made with different initial gas pressures (500, 750, and 1000 psi) in the gas reservoir.

3) A new system flow rate was set and the tests repeated with the same gas-reservoir pressures.

4) After the tests had been completed for three coolant velocities and three gas pressures, the size of the gas-discharge orifice was changed, and the test sequence in which the gas pressure and coolant velocity were varied was repeated. This was done for orifice diameters of 0.050, 0.015, and 0.016 in.

The data obtained from these tests consisted of high-speed motion pictures of the gas jet impinging on the opposite wall, transient flow rates in the channel as measured by a turbine-type flowmeter at the channel inlet, transient pressures measured by quartz, dynamic pressure transducers at the channel inlet and exit as well as at four axial locations along the channel, the transient pressure in the gas reservoir, and the static pressure at the channel inlet and exit.

The experimental procedure when the heated test section was used was basically the same as for the unheated test section, the primary difference being the added variable of the test-section power. The minimum power obtainable from the DC power supply was approximately 23 kW, which resulted in a wall heat flux of 500,000 Btu/hr-ft^2. The order in which the variables were changed differed somewhat from the unheated tests because of concern about the severe flow transients observed during the low-coolant-velocity tests in the unheated tests. Tests were performed with the various orifices and gas-plenum pressures for coolant velocities of 30 and 20 fps. Then the heat flux was increased to 1×10^6 Btu/hr-ft^2 and the tests repeated for a coolant velocity of 30 fps. Attempts to perform these tests with a coolant velocity of 10 fps resulted in destruction of the test section.

IV. EXPERIMENTAL SYSTEM

The high-pressure (2000 psi) test facility used for these tests is schematically illustrated in Fig. 1. It consists of an electrically heated test section with provisions for releasing high-pressure gas through an orifice in one wall, a gas-separation plenum, an air-cooled heat exchanger, circulation pumps, a system "makeup" pump, and a bypass flow line around the test section.

Demineralized water was circulated through the loop by Westinghouse 100-A (100-gpm) canned rotor pumps. Four of these pumps were available in the system for a combined capacity of 400 gpm. The water flow was regulated by air-operated control valves. The system pressure was maintained manually by balancing the system makeup flow against the fluid bled from the system through the discharge valve and through various gas-bleed lines in the high parts of the loop. Coolant expansion was accommodated by a 5-gal accumulator in the discharge line of the makeup pump.

A 375-kW DC power supply was used for resistance heating of the

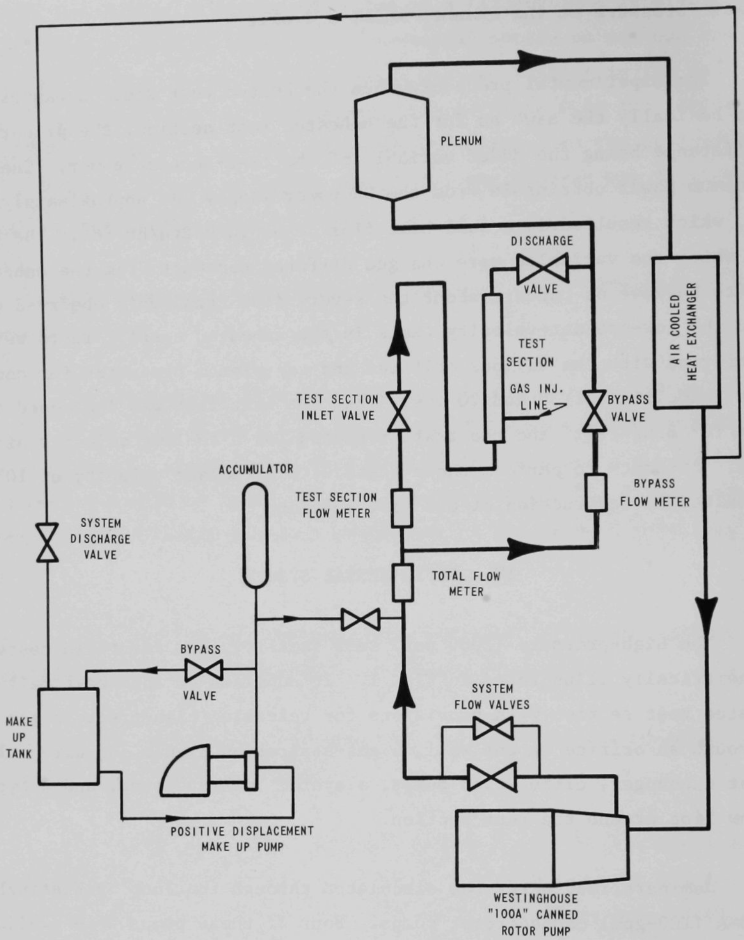

Fig. 1 Schematic of Large, 2000-psi Test Facility

test section. The output power of the rectifier was manually controlled by means of a magnetic amplifier circuit. Current transducers were used to measure the output voltage of each unit and the total voltage applied to the load. The minimum power output for the single rectifier used for these tests was 23 kW. This corresponded to a test-section heat flux of 500,000 Btu/hr-ft^2.

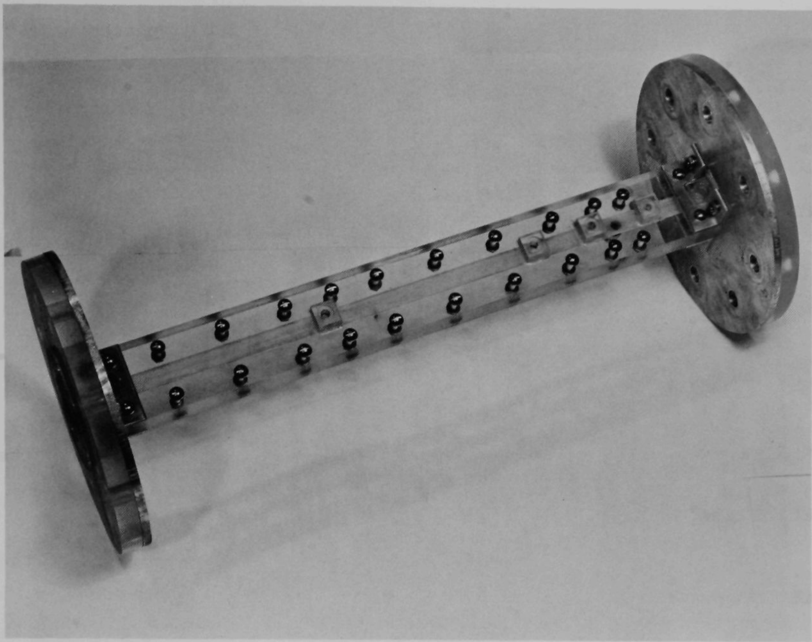

Fig. 2 Unheated Test Assembly

The unheated test section is shown pictorially in Fig. 2 and schematically in Fig. 3. The test section was made of clear Lucite to allow high-speed motion pictures to be taken of the impinging jet. The rectangular flow channel was dimensionally identical with the channel in the heated test section. The channel dimensions were 0.050 x 0.578 x 22.0 in.

Six piezoelectric dynamic pressure transducers were mounted along the length of the test section. The position of the first and last transducers corresponded to transducer positions in the heated channel. This allowed a comparison of the pressure pulses in the two test assemblies.

The gas jet was located 4 in. from the channel entrance. (This corresponds to the position of the thermocouple located furthest downstream in the heated test section.) These gas jets issued from orifice

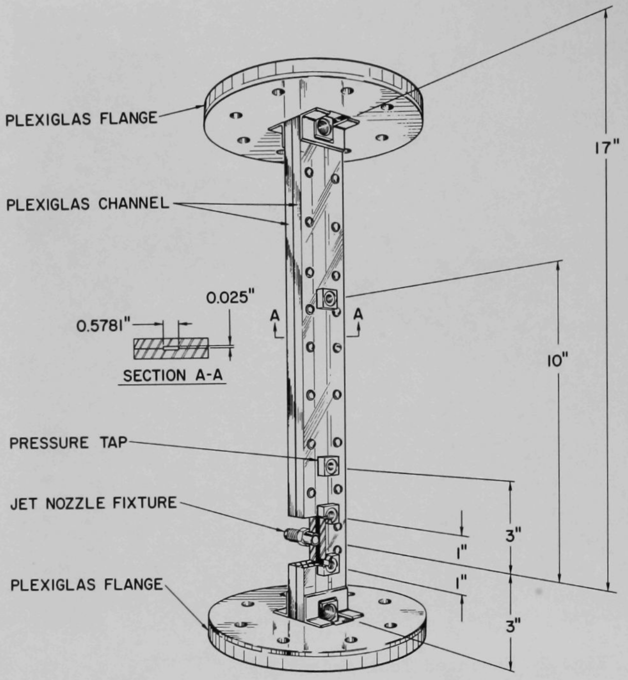

Fig. 3 Unheated Gas-release Test Section

plates brazed to the end of a short piece of 1/4-in. stainless steel tubing. The tubing was inserted into the test section through a Swagelock fitting and secured so that the orifice plate was flush with the inside wall.

The heated test section, made of copper, is shown in Fig. 4. The flow-channel dimensions were 0.050 x 0.578 x 22.0 in. The wall thickness was 0.048 in. The heated section was brazed into tapered electrodes at each end. The heated length (distance between electrodes) was 18.0 in., the total length 22.0 in. Two chromel-alumel thermocouples were welded directly to the back of the copper test section. One thermocouple was 4 in. from the inlet (2 in. from the beginning of the heated section) and directly opposite the gas orifice. The other thermocouple was 3 1/4 in. from the inlet (1 1/4 in. from the beginning of the heated region).

The heated section was carefully fitted into a "Mycalex" (glass-bonded mica) high-temperature insulator, which electrically insulates it from the massive "back-up" plates used to contain the system pressure. (The test section was originally designed for 1500 psi.) The

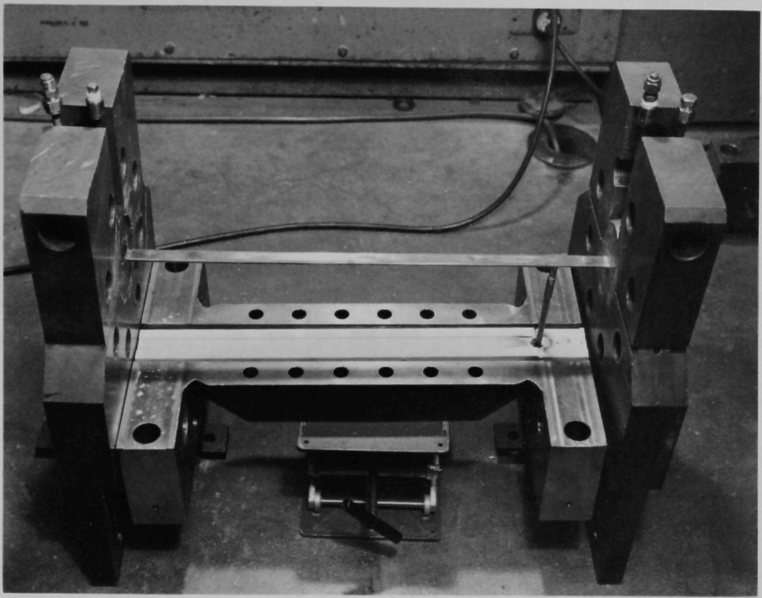

Fig. 4 Heated Test-section Assembly

tapered electrodes were clamped into bus bar connectors to complete the assembly. Figure 5 shows the end of one of the electrodes. The electrodes contained pressure taps for measurement of the inlet and outlet pressures. Chromel-alumel thermocouples in the high-pressure loop, immediately before and after the test section, gave the coolant inlet and exit temperatures.

The gas-injection system is shown in Fig. 6. The system consisted of a length of 5/8-in.-OD tubing with solenoid valves on each end. The discharge valve had a 3/8-in.-diameter orifice so that, compared to the discharge orifice, it offered minimal resistance to flow. A strain gage type of pressure transducer was inserted into the plenum to measure the transient plenum pressure.

For the unheated section, the discharge from the plenum was through

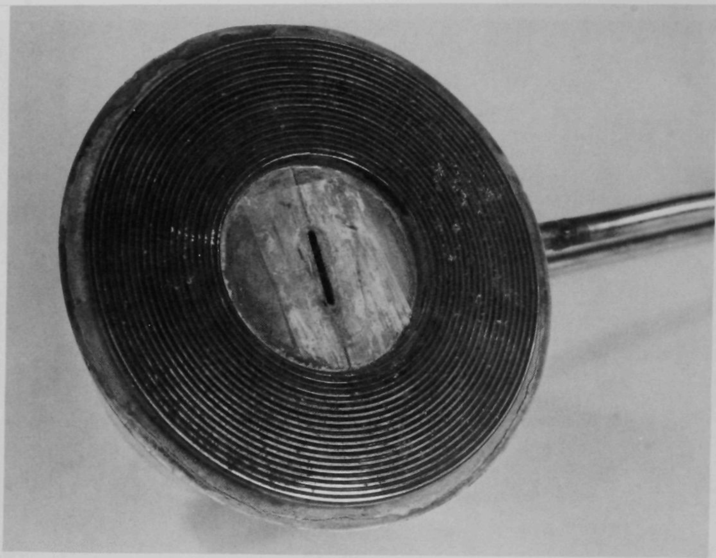

Fig. 5 End Piece (Electrode) of Heated Section

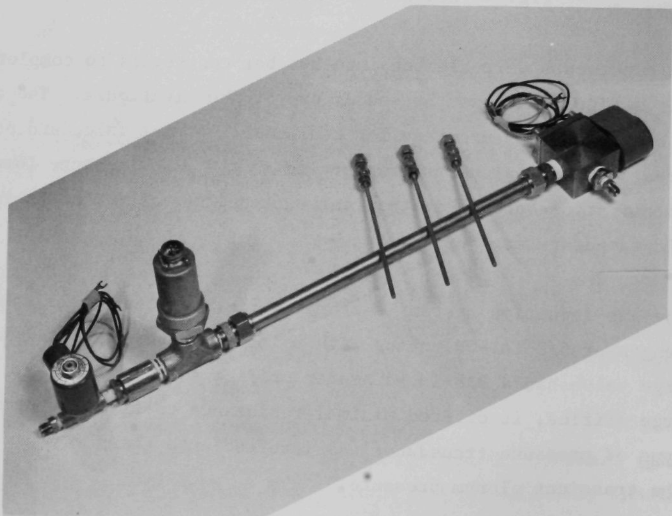

Fig. 6 Gas Plenum and Orifices

1/4-in. tubing with the orifice plate on the end. In the heated section, the discharge was through 3/16-in. tubing with orifice plates on the ends (shown in Fig. 6). These tubes were inserted through a 1/4-in. tube attached to the test section. The Swagelock connectors provided positive positioning, so the orifice was always flush with the inside wall. This arrangement allowed rapid changing of the orifice size during the test series.

The test was designed to introduce the same volume of gas into the test section that one would have in the reactor subassembly at reactor conditions of 1000°F and 3 atm. This was calculated as follows:

Assumptions:

Fuel-pin dimensions: 0.23-in. OD x 36 in. long

Cladding: 0.015 in.

Fuel-pin plenum volume: 17 cc

Fuel density: 0.85 theoretical

Burnup: 100,000 MWD/MT

Fission gas production rate: 27.9 cc (STP/MWD)

(This comes from an assumption of 26 fission gas atoms/100 fissions)

Fission gas released from fuel: 100%

$$\text{Fuel volume} = (0.785)[(0.20\text{ in.})\ (2.54\text{ cm/in.})]^2(91\text{ cm}) = 18.8\text{ cc}$$

$$\text{Weight of heavy atoms} = (18.8\text{ cc})\ (10.9\text{ gm/cc})\ (0.85)\ \left(\frac{238}{238+32}\right) = 153.7\text{ gm U-Pu}$$

$$\text{Volume of fission gas products} = (27.89\text{ cc-STP/MWD})\ \frac{(10^5\text{ MWD/MT})}{(10^6\text{ gm/MT})}\ (153.7\text{ gm}) = 428\text{ cc(STP)}$$

$$\text{Volume of gas in reactor coolant} = (428)\ \frac{(1460)}{(492)}\ \left(\frac{1}{3}\right) - 17\text{ cc} = 406.3\text{ cc.}$$

To provide 406 cc of gas in the test section at 190 psia and 130°F (approximate coolant conditions at the point of gas release), the volume of the plenum required for gas at 70°F and 1000 psia was

$$\text{Plenum volume } V_R = (406.3 \text{ cc} + V_R) \left(\frac{190}{1000}\right)\left(\frac{530}{590}\right)$$

$$= \frac{(406.3)(0.190)(0.898)}{[1 - (0.190)(0.898)]} = 83.6 \text{ cc}$$

The actual plenum volume was 88.6 cc.

V. EXPERIMENTAL RESULTS WITH THE UNHEATED TEST SECTION

High-speed motion pictures and transient pressure and flow rate data were obtained from the unheated test section for all combinations of the following parameters:

Coolant velocity	- 10, 20, 30 fps
Initial gas pressure	- 500, 750, 1000 psi
Orifice diameter	- 0.016, 0.025, 0.051 in.
Coolant inlet pressure	- 200 psia
Bypass flow ratio	- $\geq$10:1

Data were also obtained with the flow bypass closed for two coolant velocities, 10 and 30 fps, with a 0.025-in. orifice, and an initial gas plenum pressure of 750 psi.

The films were extremely useful in helping to interpret the transient temperature data from the heated tests. They also provided information on precautions that needed to be taken in running the heated tests. For example, the films showed several instances of delay and some initial pulsing in the release of the gas after the solenoid was opened. It was determined that the orifice tube filled with water during the procedure of altering the coolant velocity. The gas release was delayed while the water was being expelled. Discharging gas through the tube after the flow conditions were changed and before taking data alleviated the situation.

Figures 7 and 8, showing frames taken from two of the films,

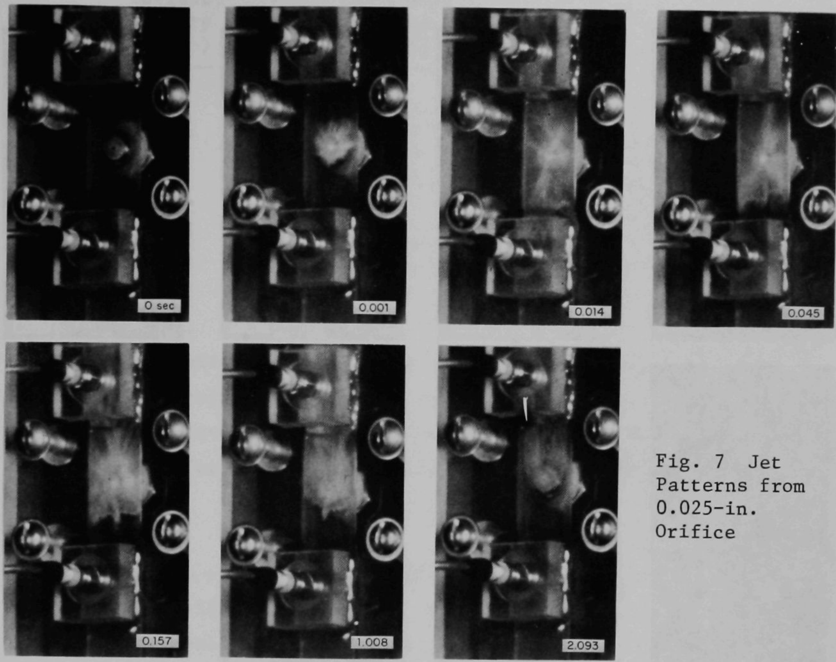

Fig. 7 Jet Patterns from 0.025-in. Orifice

illustrate the two most significant jet phenomena observed. The photographs show the coolant flow channel (located between the bolt heads), the orifice in the center of the picture, and two piezoelectric pressure transducers located 1 in. above and below the orifice. The coolant flow is in the upward direction.

Figure 7 shows conditions observed for a coolant velocity of 20 fps, an orifice diameter of 0.025 in. and an initial gas plenum pressure of 1000 psi. Points of interest are shown at t = 0.014 sec, which illustrates the maximum downward travel of the gas was to the lower transducer, approximately 1 in. The frame at t = 0.045 sec shows the "stable" interface that developed about 1/2 in. below the orifice. The "rays" of spray deflecting from the point of impingement may be seen in several of the pictures.

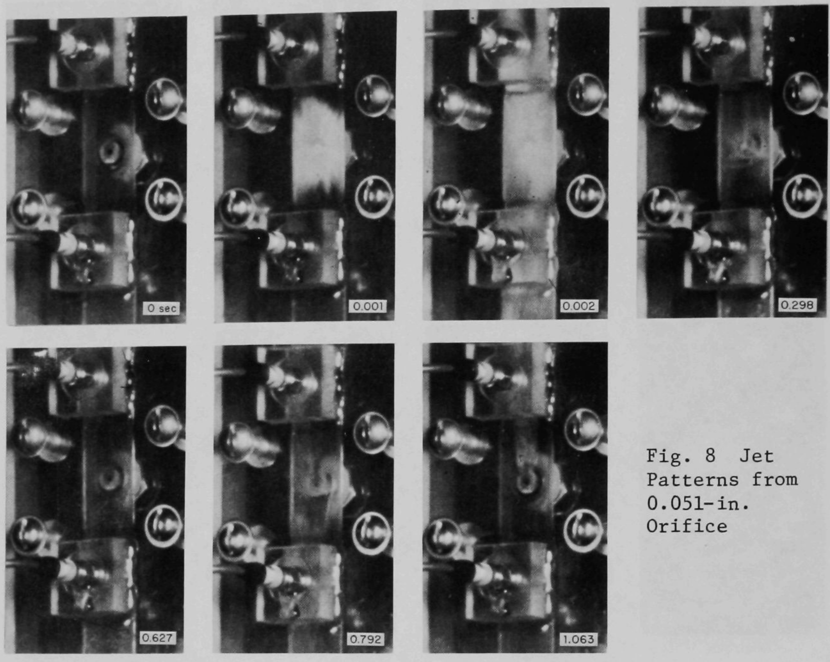

Fig. 8 Jet Patterns from 0.051-in. Orifice

Figure 8 shows conditions observed for the same test parameters as above for an orifice diameter of 0.051 in. In this series, the coolant was completely expelled from the channel. The pictures show that spray still existed about 0.3 sec into the transient. The frame at t = 0.627 sec illustrates almost complete dryout of the channel. The instant that coolant flow re-enters the channel occurs about 0.8 sec after initiation of the transient. Flow recovery times for all of the tests with the 0.051-in. orifice are tabulated below.

Coolant Velocity (ft/sec)	Gas-plenum Pressure (psi)	Flow Recovery Time (sec)
10	500	0.25
20	500	0.21
30	500	0.14
10	750	0.65
20	750	0.55
30	750	0.33
10	1000	0.89
20	1000	0.80
30	1000	0.54

The flow patterns observed for the 0.016-in. orifice were relatively unspectacular. The maximum downward motion of the gas was about 1/4 in. The spray pattern did not even completely fill the width of the channel except for the highest pressure and the lowest coolant flow rates.

For a given orifice size the pressure pulses measured 1 in. upstream of the orifice increased as the gas-plenum pressure was increased. The observed range of pressure pulses was 8-19 psi for the 0.016-in. orifice, 20-50 psi for the 0.025-in. orifice, and 45-150 psi for the 0.051-in. orifice. Determination of the maximum pressure pulses was difficult because the recording-galvanometer traces for the pressure transducers were overlapping and the rapid galvanometer motion produced light traces.

Tests were run with the coolant-bypass valve closed to compare the results with those for the open-bypass experiments. The bypass flow ratio was 10:1 for the 30-fps test and 30:1 for the 10-fps test. The primary effect of no bypass flow was to reduce the magnitude of the flow transient. The observed pressure pulses at the initiation of the transient were approximately the same for both flow conditions. Figure 9 shows a comparison of the transient pressures with the bypass open and the bypass closed.

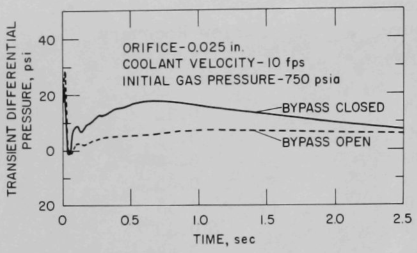

Fig. 9 Transient Pressures in Coolant Channel at 1 in. from Orifice

VI. EXPERIMENTAL RESULTS WITH THE HEATED SECTION

To prevent premature termination of the heated tests, the test parameters were varied in a manner that would cause the most severe flow transients to occur during the later stages of testing. The ranges of parameters studied were:

Heat Flux (Btu/hr-ft^2)	Coolant Velocity (ft/sec)	Orifice Diameter (in.)	Gas-plenum Pressure (psi)
500,000	30	0.016	500, 750,[a] 1000
		0.025	500, 750,[a] 1000
		0.050	500, 750,[a] 1000
	20	0.016	500, 750,[a] 1000
		0.025	500, 750,[a] 1000
		0.050	500, 750,[a] 1000
	10	0.016	500
1,000,000	30	0.016	500, 750,[a] 1000
		0.025	500, 750,[a]
		0.050	500, 750,[a]

[a]No bypass-flow tests performed for these conditions.

To ensure consistent results from the heated tests, data were obtained from at least two gas releases for each combination of variables tested. If significant differences existed in the osillograph traces, additional tests were run. In general, the reproducibility in the data from two similar tests was remarkable. Only in the final test, resulting in destruction of the test section, were there significant variations. This will be discussed later.

Data from the heated tests revealed that in every case the surface at the point of jet impingement was cooled below its steady-state temperature. Only for cases in which the channel was blown completely dry (tests with the 0.051-in. orifice) did the wall temperature rise to values that exceeded the initial steady-state value, but only after an initial temperature reduction. Examples of the data obtained from the tests are shown in Figs. 10, 11 and 12.

Figure 10 shows the transient response for a test with a 0.016-in. orifice, a coolant velocity of 20 fps, bypass flow, a heat flux of 0.5 x 10^6 Btu/hr-ft^2, and an initial gas pressure of 1000 psia. The trace labeled WALL TEMP #2 is the output of the thermocouple located directly opposite the orifice. The WALL TEMP #1 trace is from the thermocouple located 3/4 in. below the orifice. The marks labeled T_1^o, T_2^o, and w^o, indicate the initial steady-state values of the wall temperatures and the flow rate.

The temperature of the wall at the point of jet impingement may be seen to decrease rapidly from 197 to 172°F. After 2.5 sec it has risen only to 176°F. The plenum-pressure data indicate there was still about 600 psia pressure at 2.5 sec, so a strong jet was still issuing from the orifice. Although the wall directly opposite the orifice was cooled, temperature measurements 3/4 in. upstream of the jet (WALL TEMP #1) indicate the wall temperature increased from 198°F to a maximum of 241°F. This effect was observed in every transient.

Figure 11 shows the response for the same coolant velocity and plenum pressure as above, but with a 0.025-in. orifice. The plenum-pressure trace indicates that very little driving pressure remains for the jet at 2.5 sec. The flow dropped to about one-half its initial value and the lower wall temperature, T_1, increased from 190 to 379°F. The wall temperature at the jet, T_2, again showed a decrease and then some slight oscillations from 0.5 to 1.0 sec. The maximum temperature achieved during the oscillations was very close to the steady-state value. The frequency and phase of the wall-temperature oscillations corresponded closely with inlet-pressure fluctuations (maximum ΔP of about 6 psi), which indicates they are probably the result of flow oscillation. Although the flow and T_1 traces do not indicate flow oscillation for this test, there was evidence of slight fluctuations in those traces in the duplicate test run with these same conditions.

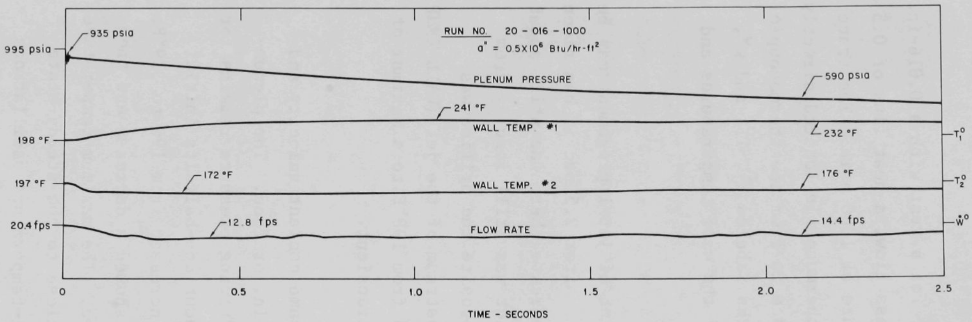

Fig. 10 Test Data for 0.016-in. Orifice

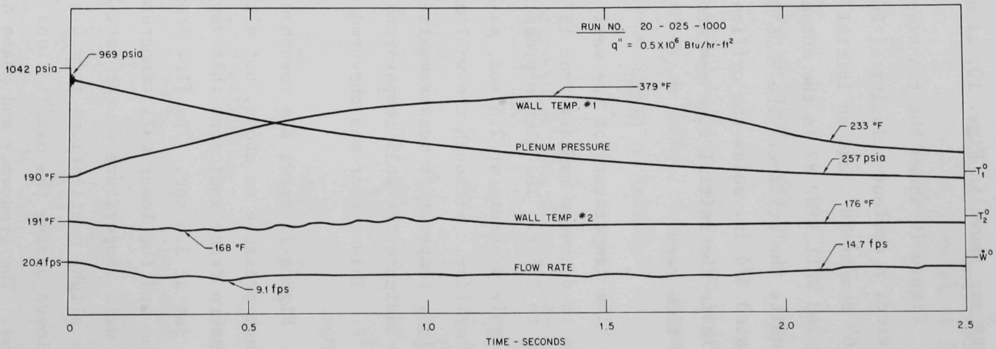

Fig. 11 Test Data for 0.025-in. Orifice

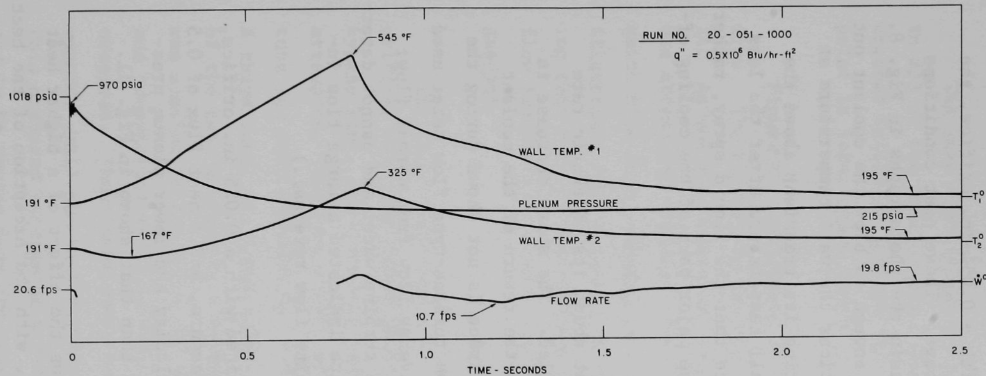

Fig. 12 Test Data for 0.051-in. Orifice

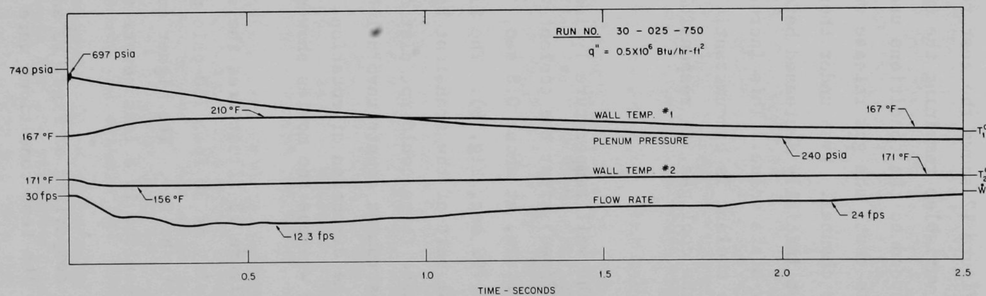

Fig. 13 Test Data for 0.025-in. Orifice and Heat Flux of 0.5×10^6 Btu/hr-ft^2

The photographs shown in Fig. 7 were taken for nearly identical conditions as the test shown in Fig. 11. Relating these pictures to the data will provide insight into the physical processes taking place.

Figure 12 shows the test results with a 0.051-in. orifice, the other variables remaining the same as above. These test conditions correspond to the conditions used to obtain the photographs in Fig. 8. In this test, the gas release was rapid enough to blow the coolant out of the channel. Even under these conditions the wall temperature at the jet initially decreased before starting its transient above the steady-state value. This increase in wall temperature after the loss of the coolant is circumstantial evidence that the liquid spray, rather than the cool gas, is responsible for the major part of the cooling effect.

The wall temperature T_1 increased at about its adiabatic rate ($\approx$600°F/sec) after the coolant was expelled. The sharp decrease in temperature, at about 0.8 sec, indicated the return of the coolant ($t = 0.792$ sec, Fig. 8). The flow-rate trace is not shown during the initial part of the transient because the turbine-type flow meter used for these tests could not distinguish reverse flow from normal flow; the instrument counts revolutions of the turbine wheel and cannot determine the direction of rotation. (The data indicated a large flow increase whereas the movies showed a definite flow reversal.)

Figure 13 illustrates the data obtained with a 0.025-in. orifice, a velocity of 30 fps, 750-psia plenum pressure, and a heat flux of 0.5 x 10^6 Btu/hr-ft^2. The higher coolant velocity and lower plenum pressure resulted in a less severe transient than that shown in Fig. 11.

A comparison of Figs. 13 and 14 shows the effect of a higher heat flux. The test conditions were the same, with the exception of the heat flux, which was 1 x 10^6 Btu/hr-ft^2 in Fig. 14. There is definite evidence of flow oscillations in this test which is presumed to be the result of boiling in the channel. There was no indication (movie or flow meter) of a flow reversal during the initial gas release, but the flow may have been reversing during the oscillations. (The flowmeter indicated rapid changes, but it could not be determined in which direction the rotor was turning.)

The same significant trend may be seen in Fig. 14 as was observed in all of the other tests. The wall temperature opposite the jet decreased initially (before flow problems start) and the transient at that point is less severe than for T_1.

Figure 15 illustrates the effect on the transient of no bypass flow. The conditions are the same as those for the test shown in Fig. 14. As would be expected, with no bypass flow the coolant transients are attenuated rapidly. The resulting wall-temperature transients are quite mild when compared to the transients in Fig. 14. This clearly illustrates the necessity of providing adequate bypass flow when studying this mode of fuel-failure propagation, for it appears that adverse flow transients are more likely to produce failure than the impinging gas jet.

The final test series in this experiment was for a 0.016-in. orifice, 500-psia plenum pressure, heat flux of 0.5×10^6 Btu/hr-ft^2, bypass flow, and a coolant velocity of 10 fps, the first 10-fps test attempted. The first test resulted in mild and well-behaved temperature and flow transients. A second test gave almost identical results. A third test, run under identical conditions, resulted in destruction of the test section by a burnout near the exit. This final transient was similar to the first two for about one second, then large pressure and temperature oscillations, with a period of about 0.25 sec, were observed. These persisted, even though the gas release had virtually ceased, until the test section was destroyed. It is postulated that this occurred because of a failure to wait for the previously expelled gas to be vented from the experimental loop. This gas made the system "spongy" and set up resonant conditions for the flow oscillations driven by the boiling process that occurred during the low-velocity transients. Similar oscillations of about the same frequency have been observed in this loop in a variety of other test conditions. This

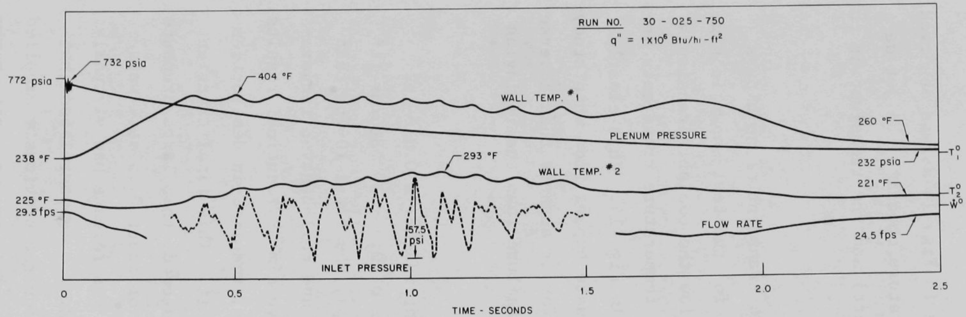

Fig. 14 Test Data for 0.025-in. Orifice and Heat Flux of 1 x 10^6 Btu/hr-ft^2

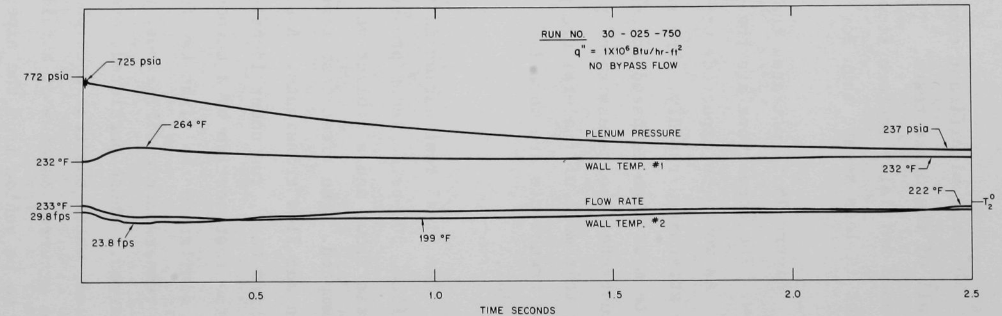

Fig. 15 Test Data for No Bypass Flow

could explain why the oscillations persisted even after the gas release was essentially completed.

The point to be emphasized from the above discussion is that the impinging gas jet did not cause failure of the test section. Failure was produced by divergent flow oscillations that caused a burnout at the test section exit.

VII. DISCUSSION OF TEST RESULTS

The high-speed motion pictures of jet impingement proved to be an invaluable aid in interpreting the results obtained with the heated section. They provided insight into the processes taking place, helped in the interpretation of the flowmeter traces, and indicated improvements to be made in the experimental techniques.

The unheated test section was instrumented to allow comparisons of the inlet and exit pressures, the coolant flow rate, and the plenum pressure with those of the heated tests. Figure 16 is an example of one such comparison and shows the coolant dynamics to be quite similar for both tests. With the exception of those heated tests influenced by coolant flow oscillations, there was generally good agreement in the coolant dynamics for both experiments.

The most significant result of the experiment was the indication of improved heat transfer at the point of jet impingement on a heated surface. Having observed the phenomenon, we need an understanding of why it happens. Two processes could produce the cooling effect: spray cooling due to entrainment of coolant droplets in the gas jet, or cooling by the high-velocity gas jet that has a low temperature produced by the sudden expansion through the orifice. It is the authors' conclusion that the dominant cooling mechanism is the result of spray cooling. The reasoning is cited below.

The high-speed motion pictures showed the existence of very forceful spray patterns on the wall of the unheated test section. These may

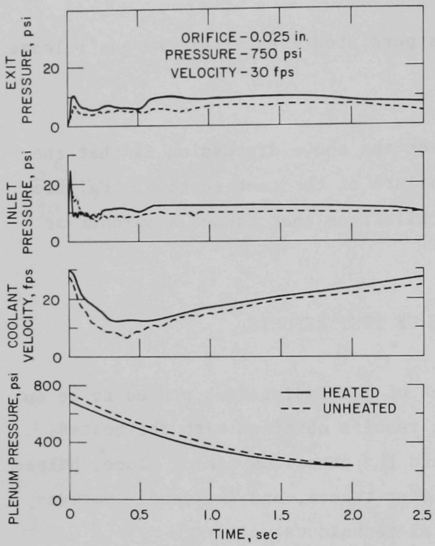

Fig. 16 Comparison of Results from Heated and Unheated Tests

be seen in Figs. 7 and 8. There is little reason to believe that similar spray patterns did not exist in the heated tests.

Comparison of the transient wall temperatures in Figs. 10, 11 and 12 reveals that, as long as coolant remained in the channel (see Figs. 10 and 11), the wall temperature was reduced at the point of jet impingement. When the coolant was blown from the channel and there was very little to be entrained in the gas jet (see Fig. 12), the wall temperature increased above its steady-state value. Figure 12 does indicate that the gas jet produces some cooling because the rate of temperature increase at the position of the jet is less than that (T_1) at the lower thermocouple. The temperature T_1 rose at a rate (≈480°F/sec) slightly less than that for adiabatic heating of the copper test section (≈600°F/sec). Heat transfer from the copper section to the cold backup plates (see Fig. 4) during the transient could account for this discrepancy. Thus, while the gas jet obviously produced some cooling, it appears liquid coolant was required in the channel to provide the major heat-transfer effect.

Finally, to obtain an estimate of the cooling capability of jets, an attempt was made to find heat-transfer correlations for jets impinging on flat plates with conditions similar to those achieved in these tests. Although nothing could be found that applied specifically to our conditions, Gardon* provided data from high-velocity jets with a

* R. Gardon, and J. Cobonpue, "Heat Transfer between a Flat Plate and Jets of Air Impinging on It," Intl. Developments in Heat Transfer, Proc. of Second Intl. Heat Transfer Conf., Boulder, Colorado, Aug 1961, pp. 454-460.

nozzle Mach No. of 0.99, orifice diameter of 0.089 in. and a ratio of plate distance to nozzle diameter of 2. The maximum Reynolds Number at the nozzle exit was 112,000.

Gardon found it necessary to express the heat-transfer coefficient at the stagnation point in terms of the "recovery" temperature and presented his results in a plot of Nusselt No. vs plate distance-nozzle diameter ratio, with parameters of Reynolds No. and orifice diameter. Extrapolating his data to the test conditions shown in Fig. 11,

Orifice Reynolds No.	=	3.6×10^5 (maximum)
Orifice Diameter	=	0.025 in.
Gas Flow Rate	=	7.38×10^{-3} lb/sec (maximum)
Plate spacing/Orifice Diameter	=	2

a maximum heat-transfer coefficient h = 3200 Btu/hr-ft^2-°F was obtained. This maximum value is 20 to 25 percent of the heat-transfer coefficient required to produce the observed results. Though it is far from definitive, this exercise again indicates spray cooling is probably the dominant mechanism.

Hindsight reveals that observing the wall-temperature transients produced by gas releases during a zero-power test, with and without coolant in the channel, would have given an indication of the gas-cooling effect.

Psuedo-heat-transfer coefficients were calculated from the heated test data by means of the equation

$$h = \frac{q''}{T_w - T_c} ,$$

where

h = heat-transfer coefficient, Btu/hr-ft^2-°F

q" = heat flux, Btu/hr-ft^2-°F (determined from test-section power input)

T_w = inner wall temperature, °F

T_c = instantaneous bulk coolant temperature, °F (calculated from test-section power and instantaneous flow rate).

These heat-transfer coefficients were best correlated with the flow rate of the escaping gas. The gas flow rate $\dot{w}$ was determined from (see the Appendix for derivation)

$$\dot{w} = -\frac{P_o V}{kRT_o}\left(\frac{1}{P^*}\right)^{\frac{k-1}{k}}\left(\frac{dP^*}{dt}\right),$$

where

P_o = initial plenum pressure (absolute)

V = plenum volume

k = specific-heat ratio

R = gas constant

T_o = initial gas temperature

P^* = dimensionless plenum pressure, $P(t)/P_o$ } obtained from data

$\frac{dP^*}{dt}$ = slope of plenum pressure trace } obtained from data

Plots of h vs $\dot{w}$ for various combinations of the parameters are shown in Figs. 17 through 23. The curves show the heat-transfer coefficient depends upon the gas flow rate to about the 0.25 power for those tests with bypass flow. For no bypass flow, the dependence is about the 0.13 power. The most common correlation in the literature, for the gas jet stagnation-point heat-transfer coefficient, shows a dependence on the orifice Reynolds Number to the 0.5 power. This would seem to imply a mechanism other than conventional gas-jet cooling.

There appears to be a trend in the data that shows a small

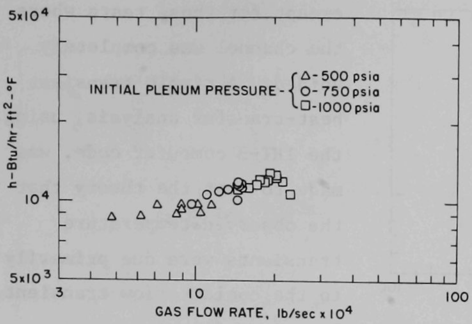

Fig. 17 Heat-transfer Coefficient for Coolant Velocity of 20 fps and Orifice Diameter of 0.016 in.

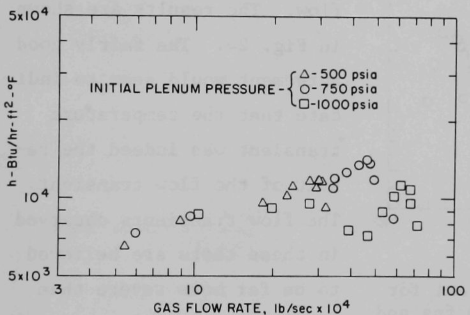

Fig. 18 Heat-transfer Coefficient for Coolant Velocity of 20 fps and Orifice Diameter of 0.025 in.

dependence on the gas flow rate when there is adequate coolant in the channel and a much stronger dependence during some of the more severe flow transients. For instance, there were very minor flow transients for the no bypass flow tests, and Fig. 21 shows a small dependence of h on gas flow rate. The data in Fig. 18 reveal h to be less than for the conditions shown in Figs. 17, 19, and 20. The conditions for Fig. 18, i.e., lower velocity and larger orifice, would result in more severe flow transients than the conditions corresponding to Figs. 17, 19, and 20. In fact, the 1000-psia data show a considerable reduction in the heat-transfer coefficient, and this test was observed to have flow problems (see Fig. 11). Figure 23 also reveals a lower heat-transfer coefficient for the 10-fps data. It is postulated that this is due to less fluid being entrained in the jet, because the decreased coolant momentum does not permit as much penetration of the gas jet.

The most severe wall-temperature transients in these tests were observed to occur at a location other than the point of jet impingement. These wall-temperature measurements were obtained 3/4 in. below the orifice. The movies showed this region to be unaffected by the gas jet

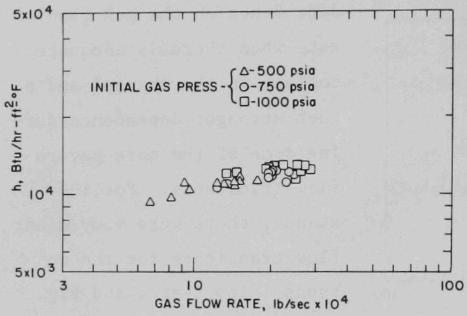

Fig. 19 Heat-transfer Coefficient for Coolant Velocity of 30 fps and Orifice Diameter of 0.016 in.

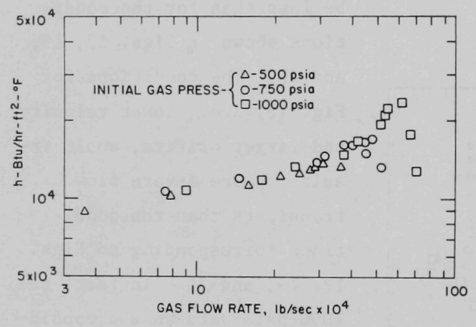

Fig. 20 Heat-transfer Coefficient for Coolant Velocity of 30 fps and Orifice Diameter of 0.025 in.

except for those tests where the channel was completely voided. A simple transient heat-transfer analysis, using the THT-B computer code, was made to test the theory that the observed temperature transients were due primarily to the coolant flow transient. Transient flow data were taken from the records and used with the Dittus-Boelter heat transfer correlation for fully developed turbulent flow. The results are shown in Fig. 24. The fairly good agreement would seem to indicate that the temperature transient was indeed the result of the flow transient. The flow transients observed in these tests are believed to be far more severe than would occur in an actual subassembly because of the very small area of the test section.

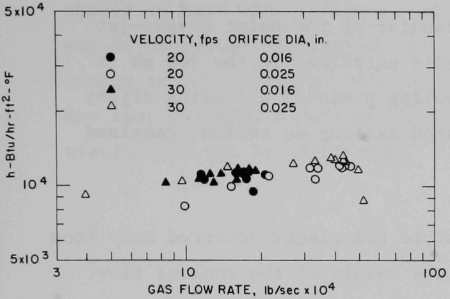

Fig. 21 Heat-transfer Coefficient for No Bypass Flow

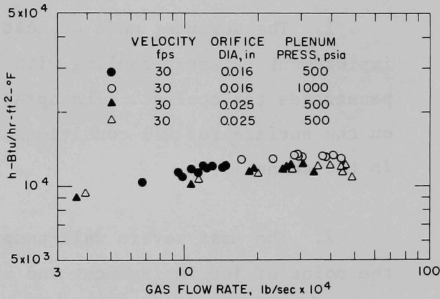

Fig. 22 Heat-transfer Coefficient for Heat Flux of 1×10^6 Btu/hr-ft^2-°F

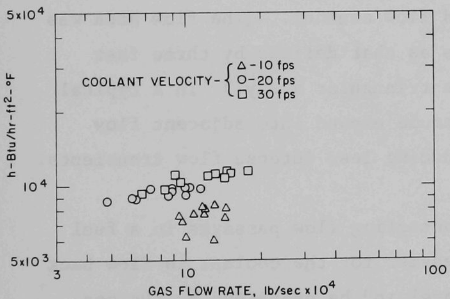

Fig. 23 Heat-transfer Coefficient for Plenum Pressure of 500 psia and Orifice Diameter of 0.016 in.

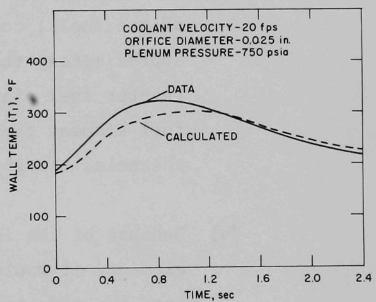

Fig. 24 Calculated Temperature Transient

VIII. CONCLUSIONS

1. The dominant mode of heat transfer at the point of gas-jet impingement is spray cooling with liquid entrained in the jet as it penetrates the coolant. The spray cooling produced a cooling effect on the surface for all conditions tested as long as coolant remained in the channel.

2. The most severe wall-temperature transients occurred away from the point of jet impingement and are the result of the coolant flow transients. The flow transients produced by a fission gas release would therefore appear to be a more probable mode of failure propagation than gas-jet impingement.

3. It is believed the transients experienced by the heated surface in these tests are more severe than would be experienced by a fuel pin in a fast reactor because of the following reasons:

(a) The flow transients were extreme in these tests because of the small, confined flow channel. (The flow area was approximately the same as that defined by three fast reactor fuel pins in a triangular array.) In a typical fuel element the gas could expand into adjacent flow channels, thereby producing less intense flow transients.

(b) Because of the interconnecting flow passages in a fuel element, it would be easier for the coolant to flow back into the affected channel and be entrained by the gas jet.

4. Since convective heat transfer to sodium is less dependent upon turbulence than is heat transfer to water, one might expect the spray cooling to be less effective in sodium. However, the spray cooling mechanism should be sufficient to prevent overheating at the point of jet impingement.

5. Since spray cooling appears to dominate, any additional testing should be done with sodium so the effect of its heat-transfer properties may be observed. Testing should also be performed in a rod-bundle geometry to integrate the effects of the flow transients into the results. Any test assembly should have adequate bypass flow so the flow transients will not be attenuated.

APPENDIX

Equations for Flow Rate

Flow rate from plenum:

$$\dot{w} = -\frac{dM}{dt} = -M_o\left(\frac{dM^*}{dt}\right) \tag{1}$$

From energy equation:

$$M^* = T^{*\left(\frac{1}{k-1}\right)} \tag{2}$$

For a perfect gas in a constant volume:

$$T^* = \frac{P^*}{M^*} \tag{3}$$

Combining equations (2) and (3):

$$M^* = P^{*\frac{1}{k}} \tag{4}$$

Substituting (4) into (1):

$$\dot{w} = -M_o\frac{d\left(P^{*\left(\frac{1}{k}\right)}\right)}{dt} = -\frac{M_o}{k}\left(\frac{1}{P^*}\right)^{\frac{k-1}{k}}\left(\frac{dP^*}{dt}\right) \tag{5}$$

Substituting for M_o:

$$\dot{w} = -\frac{P_oV}{kRT_o}\left(\frac{1}{P^*}\right)^{\frac{k-1}{k}}\left(\frac{dP^*}{dt}\right) \tag{6}$$

where:

$\dot{w}$ = gas flow rate

M = mass of gas in plenum

M_o = initial mass of gas in plenum

M^* = M/M_o

T = absolute gas temperature in plenum

T_o = initial gas temperature in plenum

T^* = T/T_o

k = C_p/C_v

P = gas pressure in plenum

P_o = initial gas pressure in plenum

P^* = P/P_o

V = plenum volume

R = gas constant

ACKNOWLEDGMENTS

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